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Thermodynamic and Economic Studies of a Combined Cycle for Waste Heat Recovery of Marine Diesel Engine

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Abstract: In the present study, the thermodynamic and economic performance of a combined thermodynamic cycle formed by an ORC and a Kalina cycle, which can simultaneously recover waste heat of exhaust gas and cooling water of marine engine, has been analyzed. Two typical marine engines are selected to be the waste heat source. Six economic indicators are used to analyze the economic performance of this combined thermodynamic cycle system with different marine engine load and under practical comprehensive operating condition of marine engine. The results of the present study show that the combined thermodynamic cycle system with R123 as organic working fluid has the best performance. The system with cis-butene has the worst economic performance. Under practical comprehensive operating conditions of ships, R123 has the shortest Payback Periods, which are 8.51 years and 8.14 years for 8S70ME-C10.5 engine and 5G95ME-C10.5 engine, respectively. Correspondingly, payback Periods of Cyclopentane are 11.95 years and 11.90 years. The above values are much shorter than 25 years which are the lifetime of a marine ship. Under practical comprehensive operating conditions of ships, the combined cycle system can provide output power which is at least equivalent to 25% of engine power. Considering that R123 will be phased out in near future, cyclopentane may be its good successor. Cyclopentane can be used safely by correct handling and installing according to manufacturer's instructions.

Keywords: marine engine, waste heat recovery, combined thermodynamic cycle, thermodynamic and economic study, R123, cyclopentane

1. Introduction

Data from the United Nations Conference on Trade and Development (UNCTAD) show ocean shipping and global ports have taken about 80% of global trade by volume and over 70% of global trade by value. Most developing countries have higher shares [1]. Maritime transport has been growing annually by around 3.1% for the past three decades [2]. Busy maritime transport and growing shipping industry [3] have accelerated energy

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consumption that may cause a polluted marine environment and increasing emission of greenhouse gases. In order to prevent these issues, International Maritime Organization (IMO) has implemented International Convention for the Prevention of Pollution from Ships (MARPOL) [4]. Furthermore, IMO has proposed the policies on energy savings, such as: Energy Efficiency Operational Indicator (EEOI), Energy Efficiency Design Index (EEDI), and Ship Energy Efficiency Management Plan (SEEMP) [4, 5]. Waste heat recovery (WHR) is a popular and promising technology which can be used for improving marine engine efficiency and reducing the emission from the marine engines [6, 7].

Diesel engine plays an important role in marine industry because of its high thermal efficiency, good economic performance, easy starting, and great adaptability. Both two-stroke and four-stroke diesel engines are used in the marine industry. Slow speed two-stroke diesel engines are usually used for the main propulsion or turning the propeller(s) of the normal ships. While four-stroke high speed diesel engines are used for providing auxiliary power of ships. Although the low-speed two-stroke marine engine has a higher thermal efficiency than general engines, a lot of energy is wasted in the form of exhaust gas, intercooler cooling water, and jacket cooling water [8]. The previous study has shown that integration of multiple waste heat sources, including exhaust gas, cylinder, and scavenge air cooling water, can improve the effect of WHR for a large marine diesel engine [9].

Compared with power turbine generation [10], thermoelectric generation [11], waste heat boiler [12], and the steam Rankine cycle [13], which have been adopted as WHR technologies for improving the energy efficiency of marine engines, the organic Rankine cycle (ORC) performs more effectively and attracts extensive attention [14–20]. Yang and Yeh have conducted a great deal of research work on the WHR of marine engine using ORC [14–16]. They used six organic working fluids in ORC to recover waste heat from cylinder jacket water of large marine diesel engines. The result shows that the order of performance from high to low is R600a, R1234ze, R1234yf, R245fa, R245ca, and R1233zd [14]. Later they optimized the thermodynamic and economic performances of an ORC system utilizing exhaust gas of a large marine diesel engine. They found the order from high to low is R245fa, R600, R600a, and R1234ze under optimal economic performance. However, in the optimal thermodynamic performance evaluations, the order from high to low is R1234ze, R600a, R245fa, and R600 [15]. They also studied the effects of the temperatures of turbine inlets and cooling water on net power output of ORC system for large marine diesel engine waste heat recovery. The results show that the order from high to low is R1234yf, R1234ze, R152a, R600a, and R245fa in the optimal thermoeconomic performance evaluation [16]. All above work is based on subcritical ORC. Furthermore, Yang has conducted some research on WHR of marine diesel engine using transcritical ORC by himself [6, 9]. He proposed three operating models of transcritical ORC system with R1234yf as working fluid to study the utilization efficiency and economic performance of exhaust gas, cylinder cooling water, and scavenge air cooling water [6]. Lubrication oil, which is the fourth waste heat source of a marine diesel engine, was added to his later research to study the economic performance of the WHR system for a marine diesel engine based on transcritical ORC. He found R236fa has the lowest optimal levelized energy cost, followed by R1234ze, R134a, R152a, R1234yf and R290 [9]. In addition to using pure working fluid, Yang used three mixtures to study the payback period of the ORC, which was used to recover the waste heat of exhaust gas from a large marine diesel engine. R600/R1234ze was found to have the shortest payback period [4]. Song et al. designed two separated ORC systems using R245fa and benzene as the working fluids to recover the waste heat from the jacket cooling water and the engine exhaust gas of a marine diesel engine [17]. Larsen et al. used one of the cylinders in a large diesel engine as an expander of ORC in order to reduce its cost. Among 104 working fluids evaluated, cyclopropane, R245fa, and R1234ze(z) are the most promising [18]. Shu et al. proposed and designed an ORC system for the WHR of exhaust from marine diesel engines based on its most typical operational condition. They reported that R123 and R365mfc are suitable candidates at heavy and light load of engine, respectively. In other words, R123 performs better in an ORC designed for container ships, while R365mfc for bulk carrier and tankers [19]. Zhu et al. presented a parametric optimization and performance analysis of an ORC for WHR of marine engine exhaust. They found R141b performs the best, followed by R113, cyclohexane, and R600a [20].

Considering the high efficiency brought by the non-isothermal phase change characteristics of the ammonia-water working fluid, the Kalina cycle was also used as the WHR technology of marine engine. Larsen et al. [21] studied a novel process layout of the Kalina cycle, called the Split-cycle, for the application to the WHR of exhaust from large marine engines. They found the design parameters of the separator, turbine, and boiler were the key for optimizing the cycle efficiency. They also found the optimized Kalina Cycle without and with reheat have a thermal efficiency of 20.8% and 21.5%, respectively. Whereas the optimized Kalina split cycle without and with reheat have a thermal efficiency of 22.1% and 23.2%, respectively. Moreover, Larsen et al. [22] established a combined system consisting of a large marine two-stroke low-speed diesel engine, a turbo-charger and a WHR system in which a dual pressure steam cycle, a Kalina cycle, and an ORC were compared. Their optimization results based on genetic algorithm suggest that the Kalina cycle possesses no significant advantages compared to the ORC or the steam cycle. However, Singh and Pedersen have a different opinion [23]. They think that the Kalina cycle is a good candidate for WHR of marine engine due to its merits and recent developments. Being suitable for low-temperature waste heat source applications and with higher cycle efficiencies than the steam Rankine cycle and ORC, the Kalina cycle offers better savings and reduced emissions, comparatively. If we analyze the reason for such a different conclusion, we will find that among three types of waste heat of marine engine, which are exhaust gas, cooling water, and lubricant, exhaust gas has the highest thermal quality. Therefore, when ORC and Kalina cycle are used to recover the waste heat of exhaust of marine engine respectively, ORC performs better than Kalina cycle. Kalina Cycle is more suitable for the recovery of waste heat with a lower temperature than exhaust.

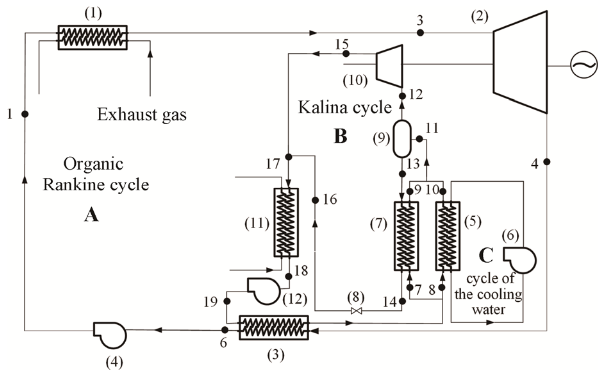
Table 1 lists some technical parameters of the ships and ship engines. It can be seen that the temperature of exhaust gas is about 250°C–500°C, up to 200°C for lubricant, and 70°C–120°C for cooling water [8]. Therefore, exhaust gas has the highest temperature and the best heat quality in waste heat of marine engine. The heat quality of cooling water is bad, but the amount of heat taken away by cooling water is very high. The lubricant temperature is a little higher than cooling water temperature, and the heat taken away by lubricant only holds a small proportion of the waste heat of the engine [27]. The fuel utilization efficiency of marine engine will be distinctly enhanced if waste heat of exhaust gas, cooling water, and lubricant is efficiently recovered through a suitable thermodynamic cycle, according to their own characteristics. Liu et al. combined steam Rankine cycle with ORC to establish a waste heat recovery system for the WHR of exhaust gas and jacket cooling water of marine engine. They found that the thermal efficiency of engine was improved by 4.42% and the fuel consumption was reduced by 9322 tons per year at an engine load of 100% [28]. On this basis, Liu et al. also combined the steam RC, ORC, and absorption refrigeration cycle for the WHR of exhaust gas and jacket cooling water of marine engine. 7620 kW electricity and 2940 kW cooling energy were obtained and the thermal efficiency of the engine was improved by 10.5% [29]. By analyzing the characteristics of waste heat of exhaust gas and cooling water of marine engine, according to the principle of “temperature matching and cascade utilization”, ORC is applicable to recovery of waste heat of exhaust gas, and the Kalina cycle to waste heat of cooling water. Therefore, following the combined thermodynamic cycle proposed for WHR of internal combustion engine in author’s previous publication [27], a combined thermodynamic cycle is used to simultaneously recover the waste heat of exhaust gas and cooling water of marine engine in this study. This combined thermodynamic cycle consists of two cycles: the organic Rankine cycle (ORC), for recovering the waste heat of high-temperature exhaust gas, and the Kalina cycle, for recovering the waste heat of low-temperature cooling water. Figs. 1 and 2 depict its system diagram and its corresponding *T-s* diagram, respectively.

In consideration of simplifying the system and controlling costs, coupled with the small proportion of the waste heat of the lubricant, compared with the cycle previously proposed by the author, the combined thermodynamic cycle used in this study does not recover the waste heat of lubricant.

Compared with the previously proposed combined energy system which only recovers the waste heat of exhaust gas [22], the combined thermodynamic cycle formed by an ORC and a Kalina cycle can simultaneously recover waste heat of exhaust gas and cooling water of marine engine. Moreover, it is a system independent of marine engine. Considering its complexity and space requirement, this combined cycle is currently more feasible for naval engines or marine engine [27]. Therefore, considering the specific

Table 1 Technical parameters of the ships and ship engines

Main diesel engine	Power	Exhaust temperature/°C	Exhaust flow	Lubricant temperature/°C
Kincaid B&W 6L90 GBE [24]	20 200 kW, 97 r/min [24]	330 [24]	110 000 $Nm^3 \cdot h^{-1}$ [24]	N/A
MAN 12K98ME-C6 [18]	68.52 MW, 104 r/min [18]	248 [18]	17.3 kg/s [18]	N/A
MAN 12K98ME/MC	68 640 kW, 94 r/min [23]	250 to 500 [8]	N/A	N/A
WARTSILA 8RT-flex68-D [25]	25 040 kW [25]	250 to 350 [26]	17.42 to 50.90 kg/s [26]	Up to 200 [26]



(1) heat exchanger for exhaust gas; (2) expander; (3) heat exchanger; (4) pump; (5) heat exchanger for cooling water; (6) pump; (7) recuperator; (8) throttle valve; (9) separator; (10) expander; (11) condenser; (12) pump.

Fig. 1 System diagram of the combined thermodynamic cycle for waste heat recovery of internal combustion engine (ICE), reprinted from Ref. [27]

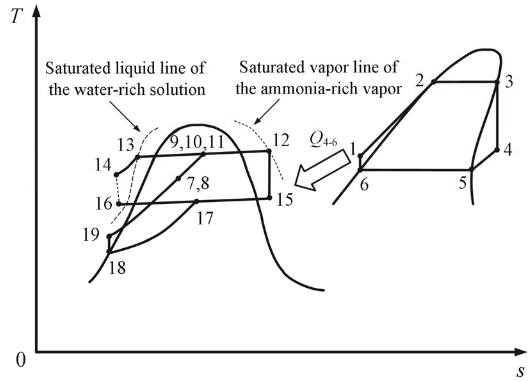


Fig. 2 *T-s* diagram of the combined thermodynamic cycle for waste heat recovery of ICE, reprinted from Ref. [27]

characteristics of waste heat of commercial marine engine, the thermodynamic performance of this combined thermodynamic cycle system is analyzed in this paper. Economic performance is an important index to investigate a thermodynamic cycle system [30–32]. The economic performance of ORC system is an important factor affecting its application and development. Not to mention the Kalina cycle system, which is more complicated than the ORC system. Therefore, the economic analysis of the combined thermodynamic cycle consisting of ORC and Kalina cycle is of great significance. Five traditional economic indicators, including total cost, net earnings, payback period, return on investment, levelized energy cost, and present value of total profit in system service life which is a relatively new indicator are used to analyze the economic performance of this combined thermodynamic cycle system. Its economic performance at practical operation condition of marine engine is obtained. The optimal load of marine engine after coupling combined thermodynamic cycle system is also obtained.

2. Modelling and Methodology

2.1 Selection and determination of waste heat source

MAN B&W and WARTSILA are two famous brands of marine diesel engines in the world. The merchant fleet around the world accounts for almost 80% of all the vessels ordered each year. Among them, 85% are powered by two-stroke diesel engines with the remainder having four-stroke engines [8]. In the market of low-speed marine engine, MAN B&W has a share of 80%, and WARTSILA has 16%. Therefore, MAN B&W engines are selected as the candidates for parameters determination of waste heat.

Tankers, bulk carriers, and container vessels are the three largest groups of vessels within the merchant fleet [33–35]. In MAN B&W's product line, today bulk carriers are often ordered with a G engine type as prime mover [34]. Meanwhile, some container ships have been ordered with two-stroke main engines with a relatively low engine speed, for example the MAN B&W super long-stroke S engine types, normally used for bulk carriers and tankers, instead of the short-stroke K engine types [35]. Based on the above consideration, two representative marine engines are selected in this paper as the supplier of waste heat of marine engines. One is 8S70ME-C10.5, which is an S-Series engine often used in container ships, and the other is 5G95ME-C10.5, which is the latest G-Series engine for bulk carriers. Specified Maximum Continuous Rating (SMCR, 100.0% of Nominal Maximum Continuous Rating, NMCR) of these two engines are 27 449 kW and 34 350 kW, respectively [36, 37]. Technical parameters of exhaust gas of engine under different loads [36, 37] are listed in Table 2. The exhaust gas is used as the heat source of the ORC in WHR system. The cooling water which has a temperature of 120°C [8] is used as the heat source of the Kalina cycle in WHR system. The flow rates of the cooling water of these two engines are 620 m^3/h and 790 m^3/h, respectively [36, 37].

2.2 Thermodynamic setting and description

Determination of all the state points in Fig. 1 and Fig. 2 are listed in Table 3.

2.2.1 Thermodynamic calculation and working fluid selection of ORC

In the ORC of combine thermodynamic cycle used for WHR of marine engine,

the net power output of expander is:

$$W_{\mathrm{ORC,\,EX}} = m_{\mathrm{w}}\left(h_3 - h_4\right) \tag{1}$$

the heat exchanged in evaporator is:

$$Q_{\mathrm{ORC,\,E}} = m_{\mathrm{w}}\left(h_3 - h_1\right) \tag{2}$$

Table 2 Technical parameters of exhaust gas of engine under different loads

Load/ % SMCR	8S70ME-C10.5				5G95ME-C10.5			
	Power/kW	Speed/ $r\cdot min^{-1}$	Exhaust gas flow/$kg\cdot s^{-1}$	Exhaust gas temperature/°C	Power/kW	Speed/ $r\cdot min^{-1}$	Exhaust gas flow/$kg\cdot s^{-1}$	Exhaust gas temperature/°C
100	27 440	91	59.5	245	34 350	80	71.6	245
80	21 952	84.5	50.9	225	27 480	74.3	61.3	225
60	16 464	76.8	40.8	228	20 610	67.5	49.1	228
40	10 976	67	28.5	255	13 740	58.9	34.3	255
20	5488	53.2	18.6	223	6870	46.8	22.4	223

Table 3 Determination of state points in the combined thermodynamic cycle used for waste heat recovery of marine engine

State point	Determination
1	s_1=s_6, evaporation pressure of ORC
2	saturated liquid state, evaporation temperature of ORC
3	saturated vapor state, evaporation temperature of ORC
4	s_4=s_3, condensation pressure of ORC
5	saturated vapor state, condensation temperature of ORC
6	saturated liquid state, condensation temperature of ORC
7,8	evaporation pressure of Kalina cycle, initial concentration of ammonia-water, and heat exchange with the condenser in ORC (heat exchanger (3))
9,10,11	initial concentration of ammonia-water, evaporation pressure of Kalina cycle, engine cooling water temperature, and terminal difference of heat exchanger
12	evaporation pressure of Kalina cycle, saturated vapor state, concentration of ammonia-rich vapor
13	evaporation pressure of Kalina cycle, saturated liquid state, concentration of water-rich solution
14	evaporation pressure of Kalina cycle, concentration of water-rich solution, energy balance in heat exchanger (7)
15	condensation pressure of Kalina cycle, s_{15}=s_{12}, concentration of ammonia-rich vapor
16	condensation pressure of Kalina cycle, h_{16}=h_{14}, concentration of water-rich solution
17	condensation pressure of Kalina cycle, initial concentration of ammonia-water, energy balance of mixing process
18	condensation pressure of Kalina cycle, initial concentration of ammonia-water, saturated state of ammonia-water
19	evaporation pressure of Kalina cycle, initial concentration of ammonia-water, s_{19}=s_{18}

the heat exchanged in condenser is:

$$Q_{\mathrm{ORC,\,C}}=m_{\mathrm{w}}\left(h_4-h_6\right) \tag{3}$$

the power consumption of pump is:

$$W_{\mathrm{ORC,\,P}}=m_{\mathrm{w}}\left(h_1-h_6\right) \tag{4}$$

In the above equations, W is power; Q is heat; m is mass flow in the unit of $kg\cdot s^{-1}$; h is enthalpy in the unit of $kJ\cdot kg^{-1}$. Subscript w stands for organic working fluid; the number stands for state point; EX stands for expander, E for evaporator, C for condenser, and P for pump.

The evaporator in ORC absorbs the waste heat of exhaust gas of marine engine, which determines the mass flow of the working fluid in ORC. Working fluid selection plays a very important role in ORC. Nowadays, HFO (Hydrofluro-Olefins) working fluids which have no ODP (Ozone Depletion Potential) and very low GWP (Global Warming Potential) are drawing more and more attention. R1234yf and R1234ze are two typical HFO working fluids that have recently been studied and used frequently. However, the critical temperatures of these two working fluids are relatively low, 367.85 K for R1234yf and 382.51 K for R1234ze(E). If they are used in this combined cycle, more fluids are needed to absorb the same amount of heat in ORC, and more heat is transmitted to Kalina cycle. Accordingly, more ammonia-water working fluid is needed in the Kalina cycle. However, engine cooling water is not enough to provide sufficient heat to heat ammonia-water working fluid for the normal operation of Kalina Cycle. Based on the above reasons, R1234yf and R1234ze(E) are not used for analysis in this paper. This result proves again that there is no ideal organic working fluid which has good thermodynamic performance, zero ODP and low GWP and can meet non-toxic, non-flammable, and non-explosive requirement [38].

In this paper, R123 and R245fa, two frequently-used working fluids during the transition period, are selected for thermodynamic and economic studies of the combined thermodynamic cycle used for WHR of marine engine. Considering cyclopentane was used for analysis in author's previous publication [27], it is still used as the working fluid in this paper. Moreover, cis-butene which has shown a good performance in subcritical cycle [39] is used as the fourth working fluid for analysis in this paper.

An ORC usually runs at subcritical rather than transcritical state. We can find a turning point on saturated vapor curve of dry or isentropic organic working fluid whose entropy value reaches the maximum ranging from normal boiling point to critical point [40]. The turning point temperature is the limit of subcritical ORC. The turning point usually has a relatively high temperature and continuing to raise the temperature of working fluid does not offer a proper way for improving thermal efficiency of a subcritical ORC with a dry or an isentropic working fluid [40]. Its more detailed explanation and application can be found in Ref. [40]. According to the above analysis, when R123, R245fa, and cis-butene are used for analysis, their evaporation temperatures are taken as the turning point temperatures, which are 423 K, 400 K, and 390 K, respectively. As for cyclopentane, its critical temperature is 511.72 K and its turning point temperature is 485 K. To avoid it working in the near critical region and too high evaporation temperature is not conducive for sufficient heat exchange with the exhaust gas, $0.9T_c$ (critical temperature) of cyclopentane is used to be its evaporation temperature, which is 460.55 K.

2.2.2 Thermodynamic calculation and parameter setting of Kalina cycle

In the Kalina cycle of combined thermodynamic cycle used for WHR of marine engine, the energy balance and mass balance equations involved in the device and process are as follows.

Heat exchanger (3):

$$Q_{\text{Kalina, }19-7,8} = Q_{\text{ORC, C}} = m_{\text{w}}\left(h_4 - h_6\right) \tag{5}$$

$$m_{\text{w}}\left(h_5 - h_6\right) = m_{\text{b}} c_p \left(T_5 - \Delta T - T_{19}\right) \tag{6}$$

$$m_{\text{w}}\left(h_4 - h_5\right) = m_{\text{b}} c_p \left(T_{7,8} - \Delta T - T_5\right) \tag{7}$$

Heat exchanger for cooling water (5):

$$n = m_{7-9} : m_{8-10} \tag{8}$$

$$Q_{(5)} = \frac{m_{\text{b}}}{n+1}\left(h_{9,10,11} - h_{7,8}\right) = m_{\text{cw}} c_p \left(T_{\text{cw,in}} - T_{\text{cw,out}}\right) \tag{9}$$

Recuperator (7):

$$n = m_{7-9} : m_{8-10} \tag{10}$$

$$m_{\text{l}}\left(h_{13} - h_{14}\right) = m_{\text{b}} \frac{n}{n+1}\left(h_{9,10,11} - h_{7,8}\right) \tag{11}$$

Throttle valve (8):

$$h_{14} = h_{16} \tag{12}$$

Separator (9):

$$m_{\text{v}} h_{12} + m_{\text{l}} h_{13} = m_{\text{b}} h_{9,10,11} \tag{13}$$

$$m_{\text{v}} + m_{\text{l}} = m_{\text{b}} \tag{14}$$

$$m_{\text{v}} C_{\text{v}} + m_{\text{l}} C_{\text{l}} = m_{\text{b}} C_{\text{b}} \tag{15}$$

Mixing process:

$$m_{\text{v}} h_{15} + m_{\text{l}} h_{16} = m_{\text{b}} h_{17} \tag{16}$$

$$m_{\text{v}} + m_{\text{l}} = m_{\text{b}} \tag{17}$$

$$m_{\text{v}} C_{\text{v}} + m_{\text{l}} C_{\text{l}} = m_{\text{b}} C_{\text{b}} \tag{18}$$

the net power output of expander (10) is:

$$W_{\text{Kalina, EX}} = m_{\text{v}}\left(h_{12} - h_{15}\right) \tag{19}$$

the heat exchanged in condenser (11) is:

$$Q_{\text{Kalina, C}} = m_{\text{b}}\left(h_{17} - h_{18}\right) \tag{20}$$

the power consumption of pump (12) is:

$$W_{\text{Kalina, P}} = m_{\text{b}}\left(h_{19} - h_{18}\right) \tag{21}$$

The total net power output of the combined thermodynamic cycle is:

$$W_{\text{tot}} = W_{\text{ORC,EX}} + W_{\text{Kalina,EX}} - W_{\text{ORC,P}} - W_{\text{Kalina,P}} \tag{22}$$

In the above equations, W is power; Q is heat; m is mass flow in the unit of $\text{kg}\cdot\text{s}^{-1}$; h is enthalpy in the unit of $\text{kJ}\cdot\text{kg}^{-1}$; C is concentration in the unit of %; c_p is specific heat capacity at constant pressure in the unit of $\text{kJ}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$; n is mass flow rate ratio; ΔT is pinch point temperature difference. Subscript w stands for organic working fluid; the number without parenthesis stands for state point; the number with parenthesis stands for device in the Kalina cycle; v stands for ammonia-rich vapor, l for water-rich solution, b for basic solution, cw for cooling water; in stands for inlet and out for outlet. EX stands for expander, E for evaporator, C for condenser, and P for pump.

The highest evaporation temperature of Kalina cycle is determined by the temperature of engine cooling water and the terminal difference of heat exchanger. The mass flow rate of ammonia-water in Kalina Cycle is determined by the condensation heat transfer in two-phase zone of ORC and the minimum heat transfer temperature difference. The temperature of the cooling water in the Kalina cycle is set to 15°C based on the ambient temperature. The initial concentration of ammonia-water is set to 82% [41]; evaporation pressure is set to 2 MPa, condensation pressure to 0.66 MPa [41]. The minimum terminal difference in heat exchanger is set to 5 K. Based on the large amount of heat carried by engine cooling water and in order to reduce the heat load of the recuperator, the value of n which stands for the mass flow rate ratio is set to 12 for R123, R245fa, and cyclopentane, 11 for cis-butene.

Based on the above thermodynamic setting and description, some parameters used for thermodynamic and economic analysis of the combined cycle with 8S70ME-C10.5 and 5G95ME-C10.5 as engines are listed in Table 4. Parameters of every state point in the combined thermodynamic cycle when R123and R245fa are used as the working fluids of the ORC are listed in Table 5. Those for cis-butene and cyclopentane are listed in Table 6.

2.3 Device design and selection

It is found that the investment in heat exchanger accounts for the largest proportion in author's previous study [42]. Shell-and-tube heat exchangers with annular fins are used in this study. Their basic parameters are selected according to reference [43] and listed in Table 7.

Convective heat transfer coefficient outside the finned tube can be calculated by [43]:

$$h_{\text{o}} = 0.3\frac{\lambda}{d_{\text{o}}}\left(\frac{G_{\max} d_{\text{o}}}{\mu}\right)^{0.625}\left(\frac{c_p \mu}{\lambda}\right)^{1/3}\left(\frac{F_{\text{f}}}{F_{\text{bo}}}\right)^{-0.375} \tag{23}$$

where, qualitative temperature is the average temperature at the inlet and outlet. Qualitative dimension is the outer diameter of tube. μ is dynamic viscosity in the unit of Pa·s; λ is heat conductivity in the unit of W/(m·K); G_{max} is the mass flow at the narrowest flow section in the unit of kg/(m^2·s); F_f is the surface area of outer fins per meter of tube, which can be calculated by:

$$F_f = 2\cdot\frac{\pi}{4}\left(d_f^{\,2}-d_o^{\,2}\right)n_f \tag{24}$$

F_{bo} is the external surface area per meter of tube, which can be calculated by:

$$F_{bo} = \pi d_o \tag{25}$$

During single-phase heat transfer process, convective heat transfer coefficient inside the tube can be calculated according to Dittus-Boelter equation by:

$$Re = ud_i/v \tag{26}$$

$$h_i = 0.023\frac{\lambda}{d_i}Re^{0.8}\,Pr^n \tag{27}$$

where, v is kinematic viscosity in the unit of m^2/s.

Table 4 Parameters of the combined cycle with 8S70ME-C10.5 and 5G95ME-C10.5 as engines

		R123	R245fa	cis-butene	cyclopentane
Condensation temperature of ORC/K		320	320	320	320
n		1:12	1:12	1:11	1:12
Inlet temperature of engine cooling water/K		393	393	393	393
Outlet temperature of engine cooling water/K	8S70ME-C10.5	321.14	317.91	319.2	327.23
	5G95ME-C10.5	325.13	322.2	318.71	330.889
Mass flow rate of engine cooling water/kg·s^{-1}	8S70ME-C10.5	172.08	172.08	172.08	172.08
	5G95ME-C10.5	219.26	219.26	219.26	219.26
Mass flow rate of cooling water for cycle condensation/kg·s^{-1}	8S70ME-C10.5	326.97	337.71	363.73	296.54
	5G95ME-C10.5	393.46	406.39	437.703	356.85
Inlet temperature of cooling water for cycle condensation/K		288	288	288	288
Outlet temperature of cooling water for cycle condensation/K		325.76	325.77	325.06	325.8

Table 5 Parameter of every state point in the combined thermodynamic cycle when R123 and R245fa are used as the working fluids of the ORC

Working Fluid		R123				R245fa			
State Point	Concentration (by mass)	T/K	p/MPa	h/kJ·kg^{-1}	s/kJ·kg^{-1}·K^{-1}	T/K	p/MPa	h/kJ·kg^{-1}	s/kJ·kg^{-1}·K^{-1}
1	N/A	320.78	2.09	249.08	1.16	320.79	2.21	263.41	1.21
2	N/A	423	2.09	366.9	1.48	400	2.21	384.66	1.54
3	N/A	423	2.09	461.02	1.70	400	2.21	486.64	1.8
4	N/A	334.81	0.19	420.62	1.70	332.41	0.31	451.45	1.8
5	N/A	320	0.19	409.63	1.67	320	0.31	438.66	1.76
6	N/A	320	0.19	247.75	1.17	320	0.31	261.94	1.21
7,8	0.82	316.55	2	354.55	1.79	316.65	2	355.04	1.80
9,10,11	0.82	387	2	1539	5.21	387	2	1539	5.21
12	0.947	387	2	1867.3	6.16	387	2	1867.3	6.16
13	0.39	387	2	420.74	1.97	387	2	420.74	1.97
14	0.39	301.57	2	21.14	0.80	301.61	2	21.31	0.80
15	0.947	339.88	0.66	1688.6	6.16	339.88	0.66	1688.6	6.16
16	0.39	301.82	0.66	21.14	0.81	301.86	0.66	21.31	0.81
17	0.82	335.76	0.66	1308.4	4.95	335.77	0.66	1308.4	4.95
18	0.82	291.37	0.66	232.53	1.40	291.37	0.66	232.53	1.40
19	0.82	291.59	2	234.46	1.40	291.59	2	234.46	1.40

Table 6 Parameter of every state point in the combined thermodynamic cycle when cis-butene and cyclopentane are used as the working fluids of the ORC

Working Fluid		cis-butene				cyclopentane			
State Point	Concentration (by mass)	T/K	p/MPa	h/kJ·kg^{-1}	s/kJ·kg^{-1}·K^{-1}	T/K	p/MPa	h/kJ·kg^{-1}	s/kJ·kg^{-1}·K^{-1}
1	N/A	320.78	1.96	99.859	0.32	320.69	2.19	–1.75	–0.014
2	N/A	390	1.96	280.31	0.83	460.55	2.19	328.83	0.83
3	N/A	390	1.96	536.92	1.49	460.55	2.19	561.03	1.33
4	N/A	373.32	0.41	523.45	1.49	350.46	0.09	428.92	1.33
5	N/A	320	0.41	465.21	1.47	320	0.09	386.3	1.208
6	N/A	320	0.41	97.234	0.32	320	0.09	–4.65	–0.014
7,8	0.82	318.62	2	364.68	1.82	317.49	2	359.14	1.81
9,10,11	0.82	387	2	1539	5.21	387	2	1539	5.21
12	0.947	387	2	1867.3	6.16	387	2	1867.3	6.16
13	0.39	387	2	420.74	1.97	387	2	420.74	1.97
14	0.39	294.95	2	–8.46	0.70	301.92	2	22.69	0.81
15	0.947	339.88	0.66	1688.6	6.16	339.88	0.66	1688.6	6.16
16	0.39	295.19	0.66	–8.46	0.71	302.16	0.66	22.69	0.81
17	0.82	335.06	0.66	1301.6	4.93	335.8	0.66	1308.7	4.95
18	0.82	291.37	0.66	232.53	1.40	291.37	0.66	232.53	1.40
19	0.82	291.59	2	234.46	1.40	291.59	2	234.46	1.40

Table 7 Basic parameters of shell-and-tube heat exchangers with annular fins

Parameter	Value
Nominal diameter of heat exchanger, D/mm	500
Material of heat exchanger	Steel
Outer diameter of heat exchange tube, d_o/mm	25
Inner diameter of heat exchange tube, d_i/mm	21
Thickness of heat exchange tube, δ/mm	2
Fin thickness, δ_f/mm	1
Fin spacing, S_f/mm	3
Fin height, H_f/mm	6
Fin diameter, d_f/mm	37
Number of fins per meter	250
Total efficiency of fin wall, η	0.917
Finned ratio, β	5.322

Convective heat transfer coefficient of two-phase boiling inside the tube can be calculated by [44]:

$$h_{\mathrm{i}}=0.023\left[\frac{u(1-x)d_{\mathrm{i}}}{v}\right]^{0.8}\frac{\lambda Pr^{0.4}}{d_{\mathrm{i}}}\times\left[1+3000Bo^{0.86}+1.12\left(\frac{x}{1-x}\right)^{0.75}\left(\frac{\rho_{\mathrm{l}}}{\rho_{\mathrm{v}}}\right)^{0.41}\right] \tag{28}$$

$$Bo=\frac{\rho_{\mathrm{v}}}{\rho_{\mathrm{v}}+\rho_{\mathrm{l}}} \tag{29}$$

where, x is vapor quality.

Convective heat transfer coefficient of two-phase condensation inside the tube can be calculated by [45]:

$$h_{\mathrm{i}}=0.023\left[\frac{u(1-x)d_{\mathrm{i}}}{v}\right]^{0.8}\frac{\lambda Pr^{0.4}}{d_{\mathrm{i}}}\times\left[(1-x)^{0.8}+\frac{3.8x^{0.76}(1-x)^{0.04}}{Pr^{0.38}}\right] \tag{30}$$

In Eqs. (27), (28), and (30), qualitative temperature is the average temperature at the inlet and outlet. Qualitative dimension is the inner diameter of tube. u is the velocity of working fluid in tube, which is set to 2 m/s [43].

Total heat transfer coefficient of finned tube (k) can be calculated by [43]:

$$\frac{1}{k}=\frac{1}{h_{\mathrm{i}}}+\frac{F_{\mathrm{i}}}{2\pi\lambda}\ln\frac{d_{\mathrm{o}}}{d_{\mathrm{i}}}+\frac{1}{h_{\mathrm{o}}\beta\eta} \tag{31}$$

where F_i is the internal surface area per meter of tube, which can be calculated by:

$$F_{\mathrm{i}}=\pi d_{\mathrm{i}} \tag{32}$$

λ is the thermal conductivity of the steel, which is 51 800 W/(m·K).

The nominal volume V_s of separator is used to calculate the cost of separator. It is determined by the nominal diameter of separator barrel according to "Specification for oil and gas separators" [46] issued by National Development and Reform Commission of China.

The nominal diameter of separator barrel can be calculated by:

$$D_s = 0.0188\left(\frac{V_{g,max}}{u_e}\right) \quad (33)$$

where, $V_{g,max}$ is maximum volume flow rate of gas in the unit of m^3/h, which can be calculated by:

$$V_{g,max} = 3600 \times 1.35 \frac{m}{\rho_v} \quad (34)$$

In the above two equations, m is the mass flow rate of ammonia-water at the expander inlet of the Kalina cycle and its unit is kg/s. u_e is the gas flow rate in separator, which can be calculated by:

$$u_e = 0.75 K_s \left(\frac{\rho_l - \rho_v}{\rho_v}\right)^{0.5} \quad (35)$$

where, K_s is a constant whose value is 0.0512 [46]; ρ_l and ρ_v are the densities of liquid and gas separated from the separator, respectively.

2.4 Economic indicator description

The investment in ORC includes two parts, equipment investment and working fluid investment. Equipment investment includes investment in heat exchangers, expander, and pump. The working fluid investment is calculated by multiplying the quantity of working fluid that can be accommodated in the evaporator and condenser chamber with its price. The investment in the Kalina cycle includes heat exchangers, pump, expander, and working fluid. The investment in throttle valve and in the piping of ORC and Kalina cycle is ignored. The calculation method of investment in ammonia-water in Kalina Cycle is similar to that of ORC. In both ORC and Kalina cycle, the calculations of investment in heat exchangers are based on their heat exchanger areas. The calculations of investment in pump and expander are based on their capacity. The calculations of investment in separator are based on its nominal volume V_s.

The purchased cost of devices and working fluids in the combined thermodynamic cycle is given in the following equations. Its detailed discussion can be found in author's previous publication [42] and reference [47].

The purchased cost (C_{pur}) of heat exchanger can be estimated by:

$$\lg C_{pur} = K_1 + K_2 \lg A + K_3 (\lg A)^2 \quad (36)$$

The purchased cost (C_{pur}) of pump and expander can be estimated by:

$$\lg C_{pur} = K_1 + K_2 \lg W + K_3 (\lg W)^2 \quad (37)$$

The purchased cost (C_{pur}) of separator can be estimated by:

$$\lg C_{pur} = K_1 + K_2 \lg V_s + K_3 (\lg V_s)^2 \quad (38)$$

Table 8 lists the values of correction factors used for the above estimation equations of purchased cost of device.

Table 8 Correction factors for purchased cost of devices

Correction factor	Heat exchangers	Pump	Expander	Separator
K_1	4.3247	3.4771	2.7051	4.7116
K_2	–0.303	0.135	1.4398	–0.5521
K_3	0.1634	0.1438	–0.1776	0.0004

The bare module cost of working fluid is estimated by:

$$C_{BM,wf} = p_{wf} M_{wf} \quad (39)$$

$$M_{wf} = \rho_{wf} \cdot \frac{A}{N\pi d_i} \cdot \frac{N\pi d_i^2}{4} \quad (40)$$

where, p_{wf} is the unit price of working fluid. M_{wf} is the mass of working fluid and it is obtained by multiplying the volume of working fluid determined by the heat exchanger chamber and the system piping by the density of working fluid. A is the heat exchange area of each heat exchanger, and N is the tube number of each heat exchanger. The volume of working fluid in the heat exchanger can be estimated from the area of each heat exchanger. In this work, the unit price of R123 is 17.967 USD/kg, R245fa is 14.81 USD/kg, cis-butene is 71.785 USD/kg, cyclopentane is 1.55 USD/kg, and ammonia-water is 447.33 USD/kg.

2.4.1 Total cost of the combined thermodynamic system

Total cost of the combined thermodynamic system is the sum of equipment investment and working fluids investment in the combined cycle. It can be estimated by:

$$C_{tot} = C_{BM,f} + C_{BM,wf} \quad (41)$$

$C_{BM,f}$ is the total cost of heat exchanger, pump, expander, and separator in combined cycle system.

2.4.2 Net Earning (NE) of the combined system

Net Earning (NE) of the combined system can be calculated by:

$$\mathrm{NE} = p_e (W_{ex} - W_p) t_{op} \quad (42)$$

where, p_e is the unit grid price whose value is 0.1 USD/(kW·h). t_{op} is annual operation time of system. According to the operating conditions listed in Table 9 [48], and t_{op} takes 7920 hours here.

2.4.3 Payback Period (PP) of the combined system

Payback Period (PP) of the combined system can be calculated by:

$$\mathrm{PP} = \frac{\ln \dfrac{\mathrm{NE} - \mathrm{COM}}{\mathrm{NE} - \mathrm{COM} - iC_{tot}}}{\ln(1+i)} \quad (43)$$

where, cost of maintenance (COM) accounts for 1.5% of total cost of the combined system, COM=1.5%C_{tot}. Annual interest rate i=5%.

Table 9 Operating conditions of ships

Operating conditions	Time		Propulsion
	Ratio/%	Hours/h	Main engine load/%
Sea passage	40	3168	80
Ballast	20	1584	60
Slow steaming	25	1980	40
Manoeuvre	5	396	20
Port	10	792	0

2.4.4 Return on Investment (ROI) of the combined system

The return on investment (ROI) is the ratio of the Net Earning (NE) to the Total Cost of the combined system. It is estimated by:

$$\mathrm{ROI}=\frac{\mathrm{NE}}{C_{\mathrm{tot}}} \tag{44}$$

2.4.5 Levelized Energy Cost (LEC) of the combined system

Levelized Energy Cost (LEC) presents the average power generation cost per kWh. It is calculated by considering compound interest of total cost of system and the operation and maintenance costs.

$$\mathrm{LEC}=\frac{\frac{i(1+i)^n}{(1+i)^n-1}C_{\mathrm{tot}}+\mathrm{COM}}{(W_{\mathrm{EX}}-W_{\mathrm{P}})t_{\mathrm{op}}} \tag{45}$$

where, n stands for system service life. Considering the lifetime of a marine ship is longer than 25 years [33–35], here n takes 25 years.

2.4.6 Present value of total profit in system service life of the combined system

Present value of total profit in system service life is a relatively new indicator. Its definition and detailed meaning can be found in author's previous study [42]. It is calculated by:

$$C_{\mathrm{PV}}=\sum_{1}^{n=25}\frac{\mathrm{NE}-\mathrm{COM}}{(1+i)^n}-C_{\mathrm{tot}} \tag{46}$$

where, C_{PV} is the present value of total profit in system service life; $(1+i)^{-n}$ is the formula used to calculate the present value; n is system service life of 25 years; i is annual interest rate which is 5%.

3. Results and Discussion

3.1 Performance of the combined cycle for 8S70ME-C10.5 engine

Fig. 3 depicts the comparison of different organic working fluids used in the combined thermodynamic cycle for WHR of 8S70ME-C10.5 engine at different load. The detailed values of different economic indicators can be found in Supplementary Table 1 in Supplementary Data. Thermodynamic performance of the combined thermodynamic cycle system used for WHR of 8S70ME-C10.5 engine with different organic working fluids at different load is listed in Supplementary Table 2 in Supplementary Data.

From the above figure, it can be seen that when marine engine is at full load, the combined thermodynamic cycle system used for WHR with R123 as working fluid in ORC has the best economic performance on all six indicators. When the marine engine is not at full load, the system with R123 has the best performance on five indicators, which are Total Cost, Payback Period, Return on Investment, Levelized Energy Cost, and Present Value of Total Profit in System Service Life. Only Net Earning of the system using R123 as working fluid is slightly lower than that using R245fa as working fluid. Taken together, we may conclude that the combined thermodynamic cycle with R123 as organic working fluid has the best performance. The system with cis-butene has the worst economic performance on all six indicators. If we compare cyclopentane with R245fa, the system using cyclopentane as working fluid has a better performance on four indictors, which are Total Cost, Payback Period, Return on Investment, and Levelized Energy Cost. While the system with R245fa has a better performance on two indictors, which are Net Earning and Present Value of Total Profit in System Service Life.

When the marine engine is at 80% load, the WHR system using R123, R245fa, and, cyclopentane has the shortest Payback Period, the lowest Levelized Energy Cost, and the highest Return on Investment. While with cis-butene as working fluid, the above three indicators are optimal when the marine engine is at 60% load. The remaining three indicators, which are Total Cost, Net Earning, and Present Value of Total Profit in System Service Life, decrease with the decrease of engine load. In other words, the Total Cost is the lowest when the marine engine is at 20% load; the Net Earning and Present Value of Total Profit in System Service Life has the highest value when the marine engine is at 100% load.

3.2 Performance of the combined cycle for 5G95ME-C10.5 engine

Fig. 4 depicts the comparison of different organic working fluids used in the combined thermodynamic

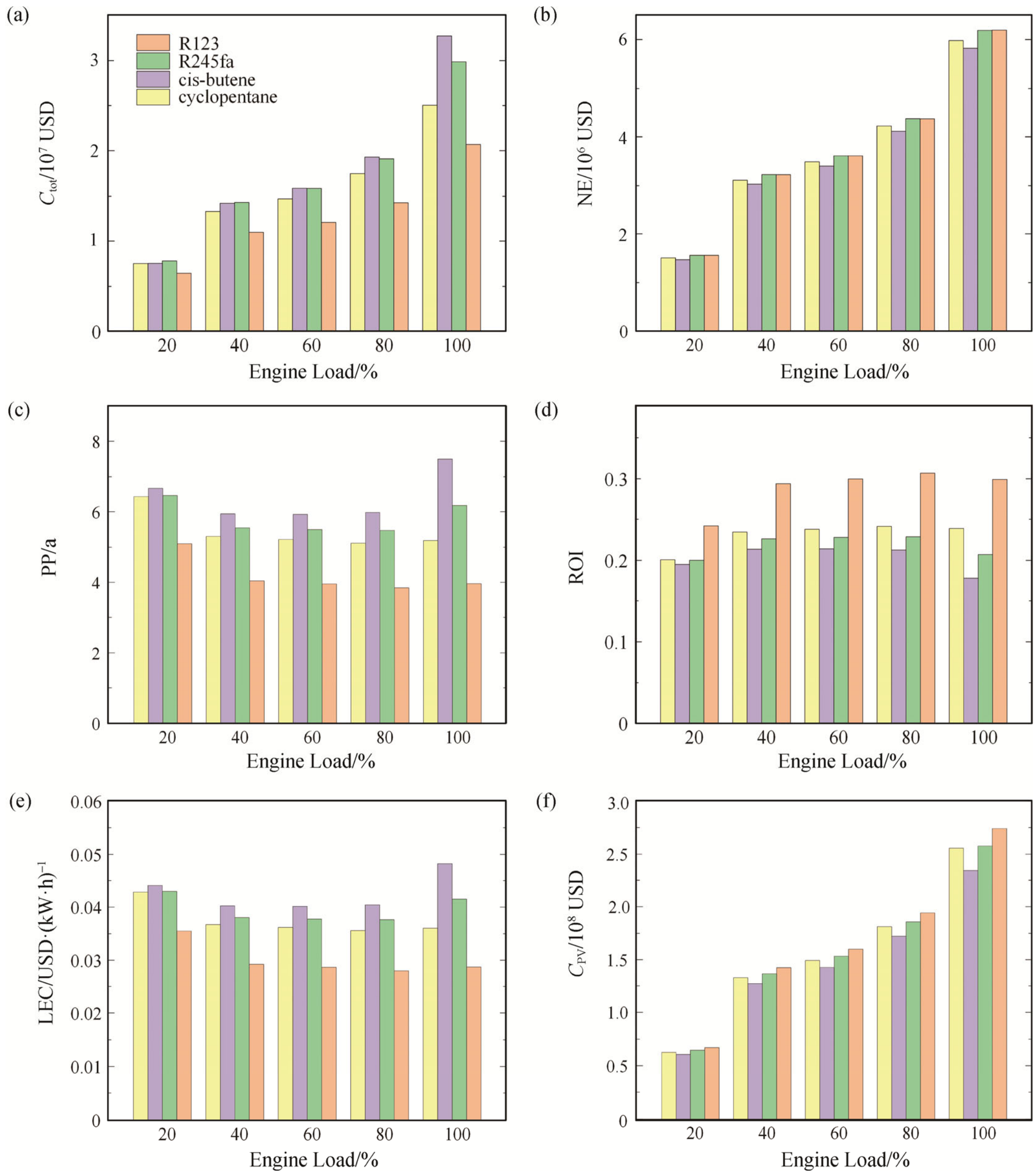


Fig. 3 Comparison of different organic working fluids used in the combined thermodynamic cycle for WHR of 8S70ME-C10.5 engine at different load, (a) Total cost, (b) Net Earning (NE), (c) Payback Period (PP), (d) Return on Investment (ROI), (e) Levelized Energy Cost (LEC), and (f) Present Value of Total Profit in System Service Life

cycle for WHR of 5G95ME-C10.5 engine at different load. The detailed values of different economic indicators can be found in Supplementary Table 3 in Supplementary Data. Thermodynamic performance of the combined thermodynamic cycle system used for WHR of 5G95ME-C10.5 engine with different organic working fluids at different load is listed in Supplementary Table 4 in Supplementary Data.

From the above figure, it can be seen that when marine engine is at full load, the combined thermodynamic cycle system used for WHR with R123 as working fluid in ORC has the best economic performance on all six indicators. When the marine engine is not at full load, the system with R123 has the best performance on five indicators, which are Total Cost, Payback Period, Return on Investment, Levelized Energy Cost, and Present Value of Total Profit in System Service Life. Only Net Earning of the system using R123 as working fluid is slightly lower than that using R245fa as working fluid. Taken together, we may conclude that the combined thermodynamic cycle with R123 as organic working fluid has the best performance. The system with cis-butene has the worst economic performance on all six indicators. If we compare cyclopentane with R245fa, the system using cyclopentane as working fluid has a better performance on four indictors, which are Total Cost,

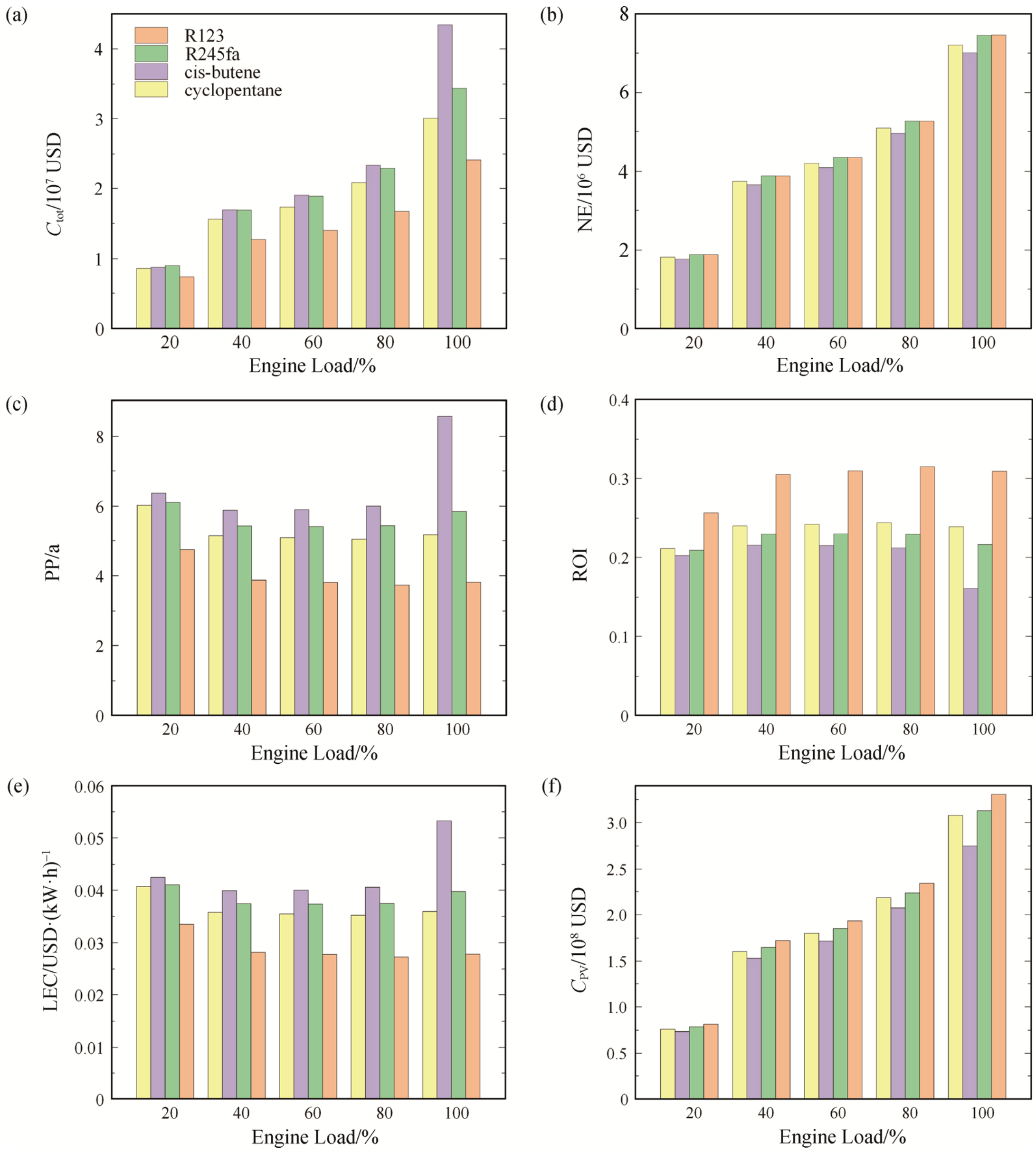


Fig 4 Comparison of different organic working fluids used in the combined thermodynamic cycle for WHR of 5G95ME-C10.5 engine at different load, (a) Total cost, (b) Net Earning (NE), (c) Payback Period (PP), (d) Return on Investment (ROI), (e) Levelized Energy Cost (LEC), and (f) Present Value of Total Profit in System Service Life

Payback Period, Return on Investment, and Levelized Energy Cost. While the system with R245fa has a better performance on two indictors, which are Net Earning and Present Value of Total Profit in System Service Life.

When the marine engine is at 80% load, the WHR system using R123 and cyclopentane has the shortest Payback Period, the lowest Levelized Energy Cost, and the highest Return on Investment. While with R245fa and cis-butene as working fluid, the above three indicators are optimal when the marine engine is at 60% load. The remaining three indicators, which are Total Cost, Net Earning, and Present Value of Total Profit in System Service Life, decrease with the decrease of engine load. In other words, the Total Cost is the lowest when the marine engine is at 20% load; the Net Earning and Present Value of Total Profit in System Service Life has the highest value when the marine engine is at 100% load.

3.3 Performance analysis of the combined cycle under practical comprehensive operating conditions of ships

According to the practical comprehensive operating conditions of marine ship listed in Table 9 [48], the performance of the combined cycle system is analyzed.

Here, the lifetime of a marine ship is taken 25 years [33–35] and the Total Cost of the system at 100% engine load is used for analysis. The results are listed in Table 10.

From Table 10, it can be seen that the combined cycle system with R123 as working fluid has the best economic performance. Under practical comprehensive operating conditions of ships, the system with R123 has the lowest Total Cost, the highest Net Earning, the shortest Payback Period, the highest Return on Investment, the lowest Levelized Energy Cost, and highest Present Value of Total Profit in System Service Life. Moreover, the previous results and discussion have shown that the combine thermodynamic cycle system with R123 as working fluid used for WHR has the best economic performance at different engine load. From Table 9, the operating condition of sea passage which has an 80% engine load accounts for a high proportion (40%) of annual operation time of marine ship engine. Under 80% engine load operating condition, the combined cycle system with R123 as working fluid still has the best economic performance. Taken together, we may conclude that the combined thermodynamic cycle with R123 as organic working fluid has the best performance.

Among the remaining three working fluids, cis-butene has the worst economic performance on all six indicators, especially it has the longest Payback Periods, which are 19.83 years and 24.52 years for 8S70ME-C10.5 engine and 5G95ME-C10.5 engine, respectively. Payback Period of cis-butene (24.52 years) for 5G95ME-C10.5 engine is almost equal to the lifetime of a marine ship (25 years). Moreover, the previous results and discussion have shown that cis-butene has its own best performance when the engine is at 60% load. Taken together, we may conclude that the combined thermo- dynamic cycle with cis-butene as organic working fluid has the worst performance.

If we compare cyclopentane with R245fa, the system using cyclopentane as working fluid has a better performance on five indictors, which are Total Cost, Payback Period, Return on Investment, Levelized Energy Cost, and Present Value of Total Profit in System Service Life. Only Net Earning of the system using cyclopentane

Table 10 Economic performance of the combined thermodynamic cycle under practical comprehensive operating conditions of ships

Engine	Working fluid	C_{tot}/USD	NE/USD	PP/a	ROI	LEC/USD·(kW·h)$^{-1}$	C_{PV}/USD
8S70ME-C10.5	R123	20 699 059.46	3 356 242.76	8.51	0.1621	0.0287	131 934 336.3
	R245fa	29 868 880.25	3 357 375.05	14.76	0.1124	0.0415	115 928 287.6
	cis-butene	32 683 629.14	3 125 997.80	19.83	0.0956	0.0483	99 402 574.07
	cyclopentane	25 039 973.36	3 210 022.83	11.95	0.1282	0.0360	117 002 764.3
5G95ME-C10.5	R123	24 125 697.41	4 040 695.85	8.14	0.1675	0.0278	160 232 197.1
	R245fa	34 423 628.67	4 042 059.06	13.96	0.1162	0.0397	140 235 991.7
	cis-butene	43 422 762.69	3 763 496.68	24.52	0.0867	0.0533	112 538 084.8
	cyclopentane	30 065 798.29	3 864 657.31	11.90	0.1285	0.0359	141 005 012.4

Table 11 Contribution of the combined cycle system to marine ship engine under practical comprehensive operating conditions of ships

Engine	Working fluid	Thermal efficiency/%	Annual output power of the combined cycle system/kW·h	Annual output power of marine engine/kW·h	Ratio of annual output power of the combined cycle system to annual output power of marine engine/%
8S70ME-C10.5	R123	7.1284	33 562 426	119 528 640	28.1
	R245fa	6.9195	33 573 748.4		28.1
	cis-butene	6.1824	31 575 734.8		26.4
	cyclopentane	7.4338	32 424 473.3		27.1
5G95ME-C10.5	R123	7.1318	40 406 941	149 628 600	27.0
	R245fa	6.9228	40 420 599.1		27.0
	cis-butene	6.1854	38 015 118.1		25.4
	cyclopentane	7.4374	39 036 941.5		26.1

as working fluid is lower than that using R245fa as working fluid. Moreover, the previous results and discussion have shown that cyclopentane has its own best performance when the engine is at 80% load for both 8S70ME-C10.5 engine and 5G95ME-C10.5 engine. However, R245fa has its own best performance when the engine is at 80% load for 8S70ME-C10.5 engine and when the engine is at 60% load for 5G95ME-C10.5 engine. Taken together, we may conclude that cyclopentane has a better performance than R245fa.

In summary, among all four working fluids, R123 has the best economic performance under practical comprehensive operating conditions of ships, followed by cyclopentane and R245fa, and cis-butene has the worst performance.

3.4 Contribution of the combined cycle system to marine ship engine

The contribution of the combined cycle system to marine ship engine with different working fluids under the practical comprehensive operating conditions of ships is listed in Table 11.

Under practical comprehensive operating conditions of ships, the ratio of annual output power of the combined cycle system to annual output power of marine engine ranges from 25.4% to 28.1%. In other words, the combined cycle system can provide output power which is at least equivalent to 25% of engine power. The system using R245fa as working fluid has the highest annual output power and cis-butene has the lowest.

3.5 Discussion on toxicity, safety, and hazards of working fluid

Based on the above comprehensive analysis, it can be seen that R123 and cyclopentane may be two suitable working fluids used in the combined thermodynamic cycle system for WHR of marine engine.

R123 belongs to HCFC (Hydrochloroflurocarbon) refrigerants. Its ODP (Ozone Depletion Potential) is 0.012, and its GWP (100-year Global Warming Potential) is 77. It will eventually be phased out under the current schedule of the Montreal Protocol, but can continue to be used in new HVAC equipment until 2020 in developed countries, and will still be produced for service use of HVAC equipment until 2030. Developing countries can use in new equipment until 2030, and can be produced for use in service until 2040. The ASHRAE safety rating for R123 is B1, which indicates that lower levels for personal exposure are allowed in normal daily operation and service conditions.

Cyclopentane is in the class of cycloalkanes, being alkanes that have one or more rings of carbon atoms. Alkanes are saturated hydrocarbons that have zero ODP and a very low GWP. Cyclopentane has a GWP of approximately 10. Hydrocarbons have excellent thermodynamic properties together with suitable physical and chemical properties which are particularly energy-efficient [49, 50]. The ASHRAE safety rating for hydrocarbons is A3, which indicates low toxic and highly flammable. However, hydrocarbons are safe to use if handled correctly and installed according to manufacturer's instructions [51]. They do not spontaneously ignite when contact with the atmosphere [52]. Therefore, cyclopentane has been widely studied and used in ORC system [53–55].

4. Conclusion

In the present study, the thermodynamic and economic performance of a combined thermodynamic cycle formed by an ORC and a Kalina cycle, which can simultaneously recover waste heat of exhaust gas and cooling water of marine engine, has been analyzed. Two typical marine engines, 8S70ME-C10.5 for container ship and 5G95ME-C10.5 for bulk carriers, were selected as the supplier of waste heat of marine engines. Five traditional economic indicators, including Total Cost, Net Earnings, Payback Period, Return on Investment, Levelized Energy Cost, and Present Value of Total Profit in System Service Life which is a relatively new indicator were used to analyze the economic performance of this combined thermodynamic cycle system with different marine engine load and under practical comprehensive operating condition of marine engine.

The results support the following conclusions:

(1) For both 8S70ME-C10.5 engine and 5G95ME-C10.5 engine, the combined thermodynamic cycle with R123 as organic working fluid used for WHR has the best performance at different engine load. The system with cis-butene has the worst economic performance.

(2) Under practical comprehensive operating conditions of ships, the combined cycle system with R123 as working fluid still has the best economic performance.

(3) Under practical comprehensive operating conditions of ships, R123 has the shortest Payback Periods, which are 8.51 years and 8.14 years for 8S70ME-C10.5 engine and 5G95ME-C10.5 engine, respectively. Correspondingly, payback Periods of Cyclopentane are 11.95 years and 11.90 years. The above values are much shorter than 25 years which are the lifetime of a marine ship.

(4) Under practical comprehensive operating conditions of ships, the combined cycle system can provide output power which is at least equivalent to 25% of engine power.

(5) Considering R123 is a transitional working fluid and will be phased out under the current schedule of the Montreal Protocol, cyclopentane may be its good successor. Cyclopentane can be used safely by correct handling and installing according to manufacturer's instructions.

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Supplementary Data

Supplementary Table 1 Economic performance of the combined thermodynamic cycle used for WHR of 8S70ME-C10.5 engine

Working fluid	Engine load/%	C_{tot}/USD	NE/USD	PP/a	ROI	LEC/USD·(kW·h)$^{-1}$	C_{PV}/USD
R123	100	20 699 059.5	6 192 757.05	3.97	0.2992	0.0287	274 081 864.1
	80	14 235 882.5	4 371 245.41	3.85	0.3071	0.0280	194 121 183.6
	60	12 059 703.2	3 615 216.58	3.96	0.2998	0.0287	160 045 985.1
	40	10 970 375.4	3 226 090.94	4.05	0.2941	0.0292	142 453 732.9
	20	6 461 247.24	1 563 570.79	5.10	0.2420	0.0355	67 037 754.0
R245fa	100	29 868 926.1	6 187 640.88	6.18	0.2072	0.0415	257 762 603.4
	80	19 135 268.1	4 372 720.13	5.47	0.2285	0.0376	185 612 824.4
	60	15 880 158.1	3 616 436.24	5.49	0.2277	0.0377	153 414 808.8
	40	14 276 550.3	3 227 179.32	5.54	0.2260	0.0380	136 716 842.9
	20	7 819 134.83	1 564 098.29	6.46	0.2000	0.0430	64 685 574.7
Cis-butene	100	32 683 629.1	5 822 076.54	7.50	0.1781	0.0483	234 512 391.2
	80	19 343 223.9	4 114 380.87	5.98	0.2127	0.0404	172 302 275.2
	60	15 885 168.9	3 402 117.77	5.92	0.2142	0.0401	142 665 792.4
	40	14 180 685.2	3 031 223.63	5.94	0.2138	0.0402	127 064 753.2
	20	7 549 172.39	1 471 834.56	6.67	0.1950	0.044 10	60 534 814.4
Cyclopentane	100	25 039 973.4	5 978 570.62	5.18	0.2388	0.0360	255 744 255.9
	80	17 508 313.7	4 224 973.07	5.12	0.2413	0.0356	181 058 647.9
	60	14 692 396.3	3 493 564.75	5.21	0.2378	0.0362	149 337 898.3
	40	13 276 883.9	3 112 701.18	5.30	0.2344	0.0367	132 731 065.4
	20	7 527 631.13	1 511 396.63	6.43	0.2008	0.0428	62 555 140.3

Supplementary Table 2 Thermodynamic performance of the combined thermodynamic cycle system used for WHR of 8S70ME-C10.5 engine

Working fluid	Engine load/%	Thermal efficiency/%	Output power/kW
R123	100	13.15%	7819.138
	80		5519.249
	60		4564.667
	40		4073.347
	20		1974.206
R245fa	100	12.75%	7812.678
	80		5521.111
	60		4566.207
	40		4074.721
	20		1974.872
Cis-butene	100	11.51%	7351.107
	80		5194.925
	60		4295.603
	40		3827.303
	20		1858.377
Cyclopentane	100	13.85%	7548.7
	80		5334.562
	60		4411.067
	40		3930.178
	20		1908.329

Supplementary Table 3 Economic performance of the combined thermodynamic cycle used for WHR of 5G95ME-C10.5 engine

Working fluid	Engine load/%	C_{tot}/USD	NE/USD	PP/a	ROI	LEC/USD·(kW·h)$^{-1}$	C_{PV}/USD
R123	100	24 125 697.4	7 452 124.45	3.82	0.3089	0.0278	331 190 666.1
	80	16 726 710.1	5 264 387.89	3.74	0.3147	0.0273	234 516 450.9
	60	14 057 133.4	4 350 665.05	3.81	0.3095	0.0278	193 402 946.1
	40	12 735 056.2	3 882 628.75	3.88	0.3049	0.0282	172 263 915.4
	20	7 328 925.53	1 883 009.98	4.75	0.2569	0.0334	81 526 041.6
R245fa	100	34 423 628.7	7 445 967.85	5.85	0.2163	0.0397	312 843 233.1
	80	22 957 850.3	5 266 163.93	5.44	0.2294	0.0375	223 690 354.7
	60	18 899 625.3	4 352 132.83	5.42	0.2303	0.0373	184 993 899.5
	40	16 912 614.0	3 883 938.63	5.44	0.2296	0.0374	165 011 722.3
	20	9 010 631.0	1 883 645.25	6.11	0.2090	0.0411	78 612 030.6
Cis-butene	100	43 422 762.7	7 006 061.85	8.57	0.1613	0.0533	275 034 224.8
	80	23 375 263.8	4 955 040.22	6.00	0.2120	0.0406	207 367 687.3
	60	19 052 657.2	4 094 215.26	5.90	0.2149	0.0400	171 800 692.9
	40	16 934 027.0	3 648 104.23	5.88	0.2154	0.0399	153 155 736.7
	20	8 768 669.17	1 772 531.95	6.37	0.2021	0.0425	73 467 604.1
Cyclopentane	100	30 065 798.3	7 194 380.78	5.17	0.2393	0.0359	307 868 955.6
	80	20 829 461.9	5 088 228.87	5.04	0.2443	0.0352	218 501 716.0
	60	17 334 664.3	4 204 265.42	5.09	0.2425	0.0354	180 325 097.9
	40	15 584 457.0	3 746 163.18	5.14	0.2404	0.0358	160 433 853.7
	20	8 616 435.37	1 820 176.59	6.03	0.2112	0.0407	76 121 910.0

Supplementary Table 4 Thermodynamics performance of the combined thermodynamic cycle system used for WHR of 5G95ME-C10.5 engine

Working fluid	Engine load/%	Thermal efficiency/%	Output power/kW
R123	100	13.15%	9409.248
	80		6646.954
	60		5493.264
	40		4902.309
	20		2377.538
R245fa	100	12.75%	9401.475
	80		6649.197
	60		5495.117
	40		4903.963
	20		2378.34
Cis-butene	100	11.51%	8846.038
	80		6256.364
	60		5169.464
	40		4606.192
	20		2238.045
Cyclopentane	100	13.85%	9083.814
	80		6424.531
	60		5308.416
	40		4730.004
	20		2298.203